



Thermodynamic analysis of a new Marnoch Heat Engine

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HIGHLIGHTS

- To develop a thermodynamic model to study the performance of a Marnoch Heat Engine (MHE).
- To investigate the effects of changing operational conditions on the MHE's efficiency.
- To assess the application of the MHE's commercial viability.

ARTICLE INFO

Article history:

Received 3 September 2012

Accepted 1 December 2012

Available online 20 December 2012

Keywords:

Heat engine
Marnoch Heat Engine
Heat recovery
Exergy
Energy
Efficiency

ABSTRACT

In this paper, recovery of waste heat from an industrial facility with a new Marnoch Heat Engine (MHE) is examined. The MHE can be operated with temperature differentials below 100 K. A flowing liquid transfers heat from the heat source into heat exchangers and then removes heat from cold heat exchangers. Compressed dry air is used as a working medium in the heat engine. In this paper, the mechanical configuration of the heat engine is presented and analyzed. A thermodynamic model is developed to study the performance of the heat engine under various operating conditions. The results show that the exergy efficiency of the MHE reaches up to 17%. The major sources of exergy loss are presented and discussed, in order to optimize the system performance.

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1. Introduction

The increases in the level of greenhouse gas emissions and fuel prices are reasons that are motivating countries to utilize various sources of renewable energy [1]. Numerous past studies have examined clean energy in place of conventional fossil fuel sources. Many of the clean renewable sources are often limited in terms of their site and reliability [2,3]. Furthermore, in many cases, the manufacturing of new energy systems, such as wind farms and nuclear power plants, is costly. Therefore, improving the performance and efficiency of current energy systems, within a competitive cost and time, is beneficial.

Waste heat recovery in industrial processes can help reduce energy consumption and lower the greenhouse gas emissions. Waste heat can be captured from the combustion exhaust gases, heated products, or heat losses from system processes [4]. Heat engines are devices that use temperature differentials to generate mechanical work; thus, they can be used as heat recovery systems.

One type of heat engine that executes the Stirling cycle is called a Stirling engine. The Stirling engine was invented by Robert Stirling in 1916 [5]. The ability of using various heat sources for power generation makes this heat engine an attractive alternative to systems that release CO₂ emissions.

Schreiber [6] tested a single cylinder free-piston Stirling engine with helium as the working fluid. The maximum power output and efficiency of this heat engine was reported to be 1.5 kW and 33%. Azetsu et al. [7] developed a laboratory scale Stirling engine. The temperatures of the heat source and sink supplied to the heat exchanger were 280 and 973 K, respectively. The working gases, which were examined in the engine, were helium, neon and argon at a mean pressure of 2.65 MPa. This laboratory scale Stirling engine produced approximately 11 kW. Abdullah et al. [8] examined the design of a double-acting Stirling engine operating with a temperature differential of 50 K. The heat source temperature in the heat engine was about 342 K and the heat sink temperature was 292 K. Cinar et al. [9] studied a beta-type Stirling engine with a total 192 cc swept volume. The maximum power output from the heat engine was 5.98 W at 208 RPM with a heat source temperature of 1273 K.

Minassians and Sanders [10] developed a new type of multiphase heat engine. This type of heat engine provided an alternative for

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Nomenclature

A	area [m ²]
C	thermal heat capacity [J/K]
C_v	specific heat at constant volume [J/kg K]
E	energy [J]
Ex	exergy [J]
$\dot{E}$	energy flow rate [W]
$\dot{Ex}$	exergy flow rate [W]
f	fouling thermal resistance [m ² K/W]
H	enthalpy [J]
h	specific enthalpy [J/kg]
k	thermal conductivity of material [W/m K]
l	length [m]
m	mass [kg]
$\dot{m}$	mass flow rate [kg/s]
N	number of experiments
Q	heat flow [J]
r	radius [m]
R	thermal resistance [m ² K/W]
T	temperature [K]
t	time [s]
U	overall heat transfer coefficient (W/m ² K)
W	work [J]

Greek letters

Δ	difference
Γ	exergetic temperature factor [dimensionless]
σ	standard deviation
η	efficiency [%]

Subscripts

C	cold
c	Carnot
cl	clean
$conv$	convection
e	electric
f	fouled
fa	from air
fi	final
H	hot
HE	heat exchanger
id	ideal
in	inlet
ini	initial
ins	inside
out	outlet
$outs$	outside
ss	stainless steel
ta	to air
th	thermal
0	reference state

Acronyms

B	Bias
MHE	Marnoch Heat Engine
PL	Precision Limit
PLC	Programmable Logic Controller
PM	Permanent Magnet
U	Uncertainty
W/G	Water and polypropylene glycol mixture

a single-phase alpha heat engine, to utilize solar energy as the heat source. The heat source temperature for the heat engine lies in the range of 398–423 K and the heat sink temperature is about 298 K [11]. Tan et al. [12] reported a maximum efficiency of 0.05% and a maximum power output of 0.01024 W for a temperature difference of 70 K from a gamma-type Stirling engine. Kang et al. [13] designed and tested a gamma-type heat engine. They reported a maximum efficiency of 2.9% at 849 K heat source temperature. The engine speed and power output at this temperature were reported as 412 RPM and 6.47 W, respectively. Minassians and Sanders [14] introduced a new type of Stirling engine using solar thermal energy to produce electricity. This new heat engine uses low-cost materials and manufacturing methods. The target concentrator collector temperature range is between 393 and 423 K. This temperature range is achievable with optical concentrators.

In this study, a new Marnoch Heat Engine (MHE) is analyzed to recover heat from low temperature differentials (less than 100 K). Energy and exergy analyses are performed to study the system performance under various operational conditions. Effects of increasing the initial charging pressure on the pressure differentials, between the hot and cold heat exchangers, are studied.

2. System description

A prototype of the MHE was designed and built in the Clean Energy Research Laboratory (CERL) at UOIT (see Fig. 1). In order to simulate various operating conditions, a dry cooler and two water heaters were used. The minimum W/G temperature that can be obtained from the dry cooler exists at the ambient temperature and the maximum liquid temperature output of the heaters is 358 K.

This assembly of the MHE has four shell and tube heat exchangers. Shells of the heat exchangers are made of carbon steel,

while tube bundles are stainless steel. Detailed information of the shells and tubes is available in Tables 1 and 2. The heat transfer liquid that flows through the system is a Water/Glycol (W/G) mixture with a 50% polypropylene glycol concentration. Each heat exchanger has one gas inlet/outlet, one liquid inlet and one liquid outlet. The liquid that flows through the hot and cold heat exchanger tubes of the MHE stays in a closed loop.

Heat is collected from the heat reservoir and transferred with W/G to the MHE heat exchangers. Compressed air inside the heat exchanger shells removes heat from the liquid. As a result, the pressure of the gas inside the shell increases. In the other heat exchangers, the liquid cools down the gas. Therefore, the gas

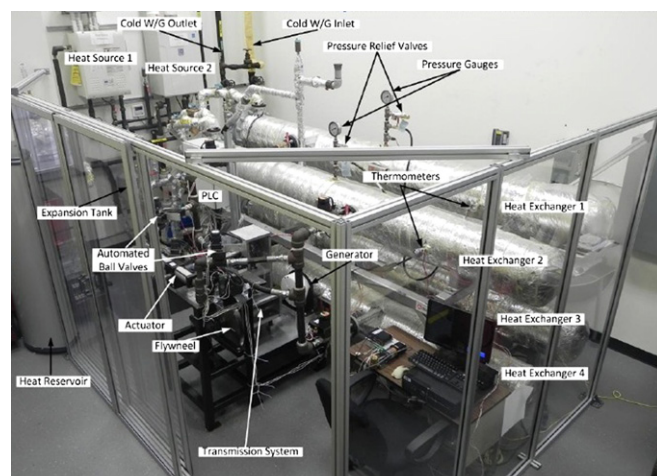


Fig. 1. Current MHE prototype in the Clean Energy Research Laboratory at UOIT.

Table 1
Shell specifications of the MHE prototype.

Total length of shell	3.62 m
Inside diameter of shell	314 mm
Outside diameter of shell	323 mm
Wall thickness of shell	9.5 mm
Total volume inside shell	255 l
Total surface area of shell	3.84 m ²
Thickness of front flanges	31.5 mm
Diameter of front flanges	485 mm
Number of bolts of front flanges	12
Diameter of bolts of front flanges	15 mm
Header length	203 mm
Applied material	SA-106 B
Thermal conductivity of alloy	53 W/m K

pressure inside the shell decreases. When the pressure differential reaches a maximum, the low pressure shell is connected to the high pressure shell. Gas moves from the high to low pressure shell. A two-way piston assembly is installed on this line. Air flow moves the piston from the high to low pressure shell. This process continues until the pressures inside the shells equalize.

Automated ball valves switch the high and low pressure sides of the piston to obtain continual strokes from the piston assembly. A diagram of the pneumatic system is shown in Fig. 2. More information about the MHE assembly and working principles are available in Refs. [15–18]. The valve motions are controlled by a PLC. For more information about the MHE control system, see Refs. [19–21]. Piston revolutions are converted to one-way rotary motion of a flywheel through a transmission system. This results in the rotation of the main flywheel. Since the flywheel stores energy, it reduces the fluctuations of the piston power output. Kinematic energy of the flywheel is converted to electricity through a PM generator. A summary of the energy conversion steps is shown in Fig. 3.

T-type and J-type thermocouples are installed on the shells to measure gas and liquid temperatures, respectively. The voltage is measured and then translated to Kelvin units for temperatures. Pressure sensors are installed on the heat exchangers to measure the air pressure inside the shell. The power output from the generator is measured by a multimeter, which can simultaneously read both voltage and current. Table 3 shows the list of measuring devices and relative errors, based on Refs. [22–25]. The inaccuracies related to the measuring devices are the bias errors. Random variations are the errors that occur due to uncontrolled parameters when making repeated measurements. These variations affect the precision of the experimental readings. Using the method of Kline and McClintock [26], the measurement uncertainty (U) for the experimental data is given by:

$$U = \sqrt{B^2 + PL^2} \quad (1)$$

Table 2
Tube bundle specifications of the MHE prototype.

Total number of tubes	66
Inside diameter of tubes	16.9 mm
Outside diameter of tubes	19.1 mm
Length of each tube	6.10 m
Total volume of tubes	59.2 l
Thickness	2.1 mm
Surface area of one tube	0.18 m ²
Total surface area of the tube bundle	12.42 m ²
Applied material	ASTM 249 TP304
Thermal conductivity of the alloy @ 323 K	13.8 (W/m K)
Weight	0.85 (kg/m)

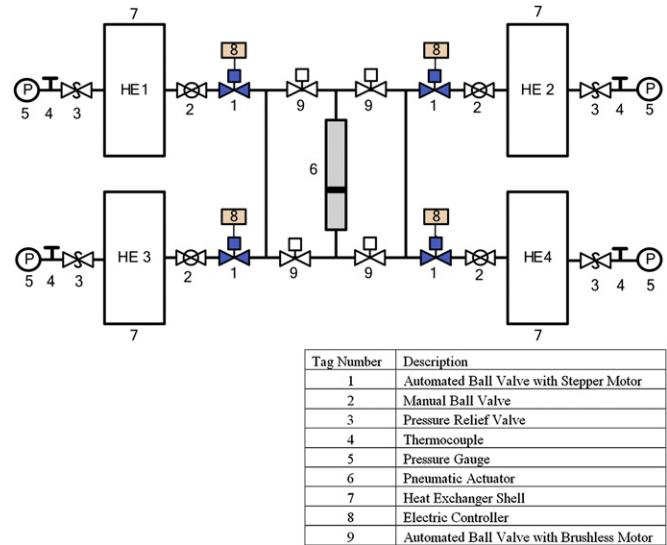


Fig. 2. Schematic of the compressed air lines and pneumatic system of the MHE.

Here B is the bias limit, and PL is the precision limit for the experimental readings. Bias is the error associated with the inaccuracy of the measuring equipment (see Table 3). The precision can be found based on the following equation [27]:

$$PL = \frac{\sigma}{\sqrt{N}} \quad (2)$$

where σ and N are the standard deviation and the number of experiments, respectively. The measurement uncertainties are

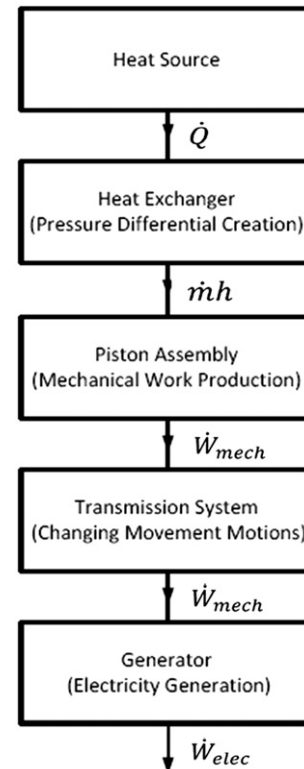


Fig. 3. Energy conversion steps in the MHE.

Table 3
Measurement devices and errors [22–25].

Equipment	Accuracy
J-type thermocouple	±0.2 K
T-type thermocouple	±1 K
Pressure sensor	±8.6 kPa
DAQ – pressure signals	±17 kPa
DAQ – temperature signals	±1 K
Multimeter – voltage	±1.8 V
Multimeter – current	±0.05 A

reported in Table 4. The resulting errors in the temperature, pressure and power measurements yield results within a 95% confidence interval.

3. Energy and exergy analyses

In this section, thermodynamic and heat transfer analyses are conducted to evaluate the performance of the MHE. Detailed information about the thermodynamic cycle of the MHE is available in Refs. [28,29]. The performance of the MHE is evaluated under different ambient conditions by imposing temperature variations on the heat source and sink. The system performance varies when the system is commissioned and executes the first cycle, when the heat engine is running continually. The MHE operation has three different modes with different initial conditions (see Table 5). Details about the continual operational modes are given as follows.

- 1) This mode starts when the amount of accumulated air inside the shell of the hot heat exchanger is higher than the cold heat exchanger, resulting from the previous operational cycle. During this mode, the heat exchanger that contains more air mass is heated and the heat exchanger with the lower air mass is cooled. Thus, the pressure differentials of continuous operation are higher than the start-up period (see Table 5).
- 2) At the second operational mode of a continuous operation, hot and cold fluids are supplied to the heat exchangers to maintain almost isothermal expansions and compressions. Since the amount of supplied/removed heat to/from the heat exchangers is constant, the process is not isothermal. The temperatures of the heat source and heat sink are constant and the liquid flow rates, through the heat exchangers, are constant. However, the transferred air mass from/into the shells is not constant. Some references e.g. [30,31], are available for more detailed analysis on transient heat and mass transfer in the MHE.
- 3) The third operational mode starts when the pressures of the compressed air inside the cold and hot heat exchangers are equal. At this point, all valves are closed and no fluid transfers into/from the heat exchangers.

The governing equations for the charging and discharging modes are transient because the flow of air mass leaving the shell and entering the other heat exchanger is time dependent. The only steady-state equation for the heat exchangers is the liquid mass

Table 4
Precision and bias errors of experimental measurements.

Variables	Precision error	Bias error
Liquid temperature	±1.0 K	±2.0 K
Ambient temperature	±1.0 K	±0.3 K
Compressed air temperature	±1.4 K	±2.2 K
Compressed air pressure	±22.8 kPa	±19.1 kPa
Power output from generator	±3.5 W	±1.8 W

Table 5

Mass and pressure variations in the heat exchangers during start-up and continuous operation.

	Mode 1		Mode 2		Mode 3
	Beginning	End	Beginning	End	Instant
<i>Start-up cycle</i>					
Air pressure	$P_H = P_C$	$P_H > P_C$	$P_H > P_C$	$P_H = P_C$	$P_H = P_C$
Air mass	$M_H = M_C$	$M_H = M_C$	$M_H > M_C$	$M_H = M_C$	$M_H < M_C$
<i>Continuous cycles</i>					
Air pressure	$P_H = P_C$	$P_H > P_C$	$P_H > P_C$	$P_H = P_C$	$P_H = P_C$
Air mass	$M_H > M_C$	$M_H > M_C$	$M_H > M_C$	$M_H = M_C$	$M_H < M_C$

balance equation, since the mass flow rate of the W/G mixture through the tube bundle is constant. Fig. 4 shows a schematic of the system that is used for the thermodynamics analysis. The following equation indicates the liquid mass balance that flows through the heat exchanger:

$$\dot{m}_{W/G_{in}} = \dot{m}_{W/G_{out}} \quad (3)$$

where $\dot{m}_{W/G_{in}}$ and $\dot{m}_{W/G_{out}}$ denote the liquid mass flow rate at the inlet and outlet of the tube bundles, respectively. The air mass at the inlet/outlet of the shell is not at steady state. During the charging mode, since the air ball valves are closed:

$$\dot{m}_{air_{in}} = \dot{m}_{air_{out}} = 0 \quad (4)$$

Here $\dot{m}_{air_{in}}$ represents the air mass flow rate that enters the shell and $\dot{m}_{air_{out}}$ is the air mass flow rate that leaves the shell. The amount of remaining gas inside the shell at any instant in time, during the operational mode, is given by

$$m_{air,fin} = \int_{t_{ini}}^{t_{fin}} \frac{dm_{air_{in/out}}}{dt} dt + m_{air,ini} \quad (5)$$

When $dm_{air_{in/out}}/dt$ is integrated over time, the result yields either the total air mass that is accumulated inside the shell or the air mass that leaves the shell.

The overall energy balance equation for each heat exchanger is given by

$$\Delta \dot{E}_{HE} = \sum \dot{E}_{in} - \sum \dot{E}_{out} \quad (6)$$

$\Delta \dot{E}_{HE}$ represents the rate of energy changes through the heat exchanger. $\dot{E}_{in}$ and $\dot{E}_{out}$ denote the energy transfer rates into and from the heat exchanger, respectively. The energy transferred from the liquid to the gas is

$$\sum_{t_{ini}}^{t_{fin}} \dot{E}_{ta} = \sum_{t_{ini}}^{t_{fin}} \dot{E}_{in} - \sum_{t_{ini}}^{t_{fin}} (\dot{E}_{out} + \dot{E}_{loss}) \quad (7)$$

where $\sum_{t_{ini}}^{t_{fin}} \dot{E}_{in}$ is the amount of energy transferred through one operational cycle. Since the stream that transfers hot liquid into the system has a constant mass flow rate and constant temperature, $\dot{E}_{in}$ is constant. The energy flow rate from the tube bundle at the outlet of the hot heat exchanger ($\dot{E}_{out}$) has a minimum value at the beginning of the charging period and maximum value at the end of this period. However, during the operation, this difference is not significant, because the heat capacity of the liquid is significantly higher than the gas. Also, $\dot{E}_{ta}$ is the energy transfer rate to the dry air inside the shell, with its maximum value at the beginning and minimum value at the end of the charging period. $\dot{E}_{loss}$ is the irreversible loss due to fluid friction and pressure drops. Since energy

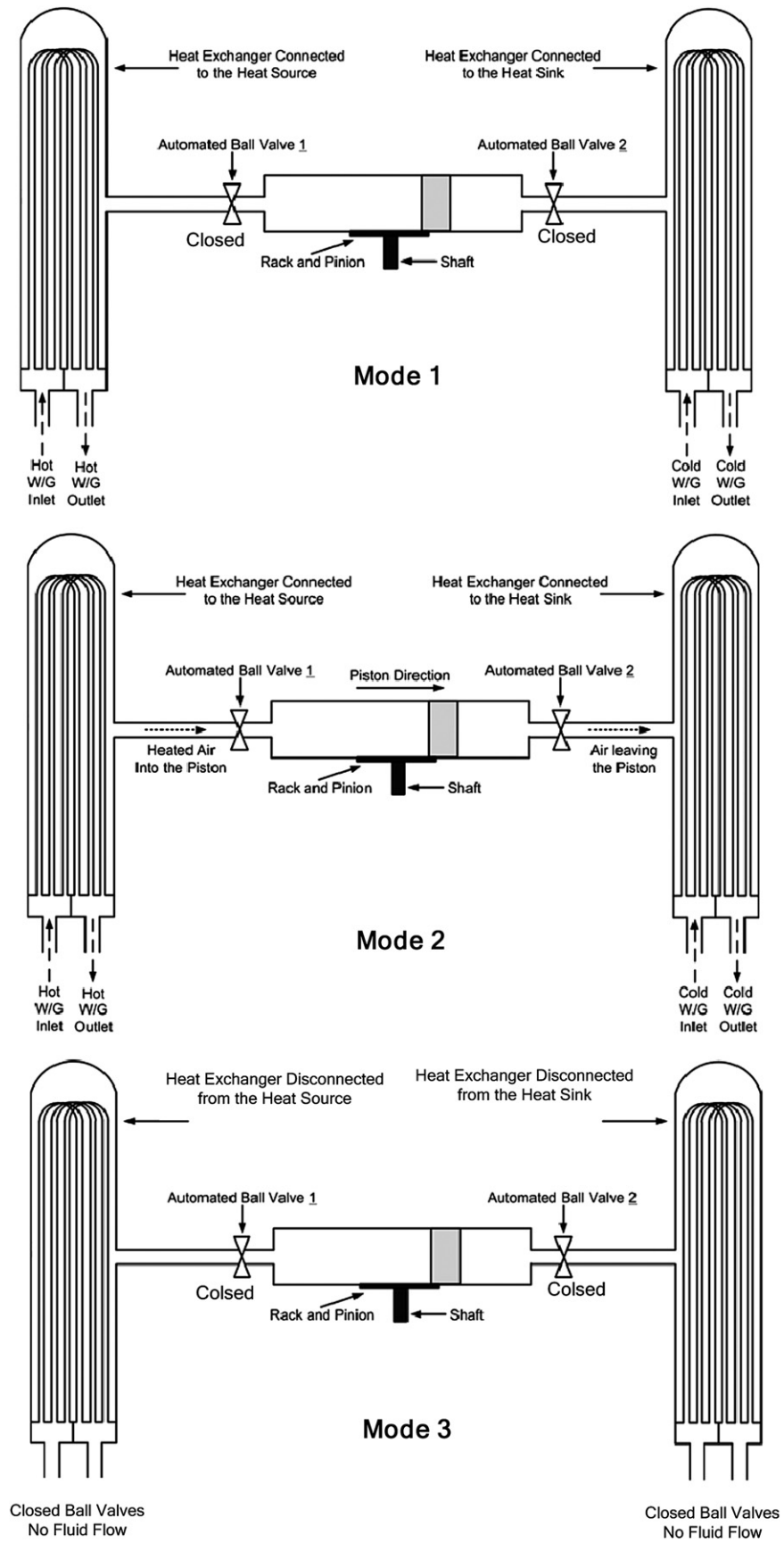


Fig. 4. Three operational modes of the MHE.

transfer occurs in the form of heat, $\dot{E}_{out}$ and $\dot{E}_{in}$ are transient and $\dot{E}_{loss}$ is assumed negligible, so Eq. (7) is written as

$$\int_{t_{ini}}^{t_{fin}} \dot{Q}_{in} dt = \int_{t_{ini}}^{t_{fin}} \dot{Q}_{out} dt + \int_{t_{ini}}^{t_{fin}} \dot{Q}_{ta} dt \quad (8)$$

Here, $\dot{Q}_{in}$ and $\dot{Q}_{out}$ are the heat transfer rates into/from the system via fluid flow through the bundles. Also, $\dot{Q}_{ta}$ is the heat transfer rate to the air and it is defined as

$$\dot{Q}_{ta} = UA_{tube} \Delta T \quad (9)$$

where U is the overall heat transfer coefficient that is determined by a generalized thermal resistance network as follows

$$U = \frac{1}{R_{convW/G} + f_{W/G} + R_{condtb} + f_{air} + R_{convair}} \quad (10)$$

Here, $R_{convW/G}$, R_{condtb} and $R_{convair}$ for the pipe are defined as

$$R_{convW/G} = \frac{1}{h_w 2\pi r_{ins} L} \quad (11)$$

$$R_{condtb} = \frac{\ln(r_{outs}/r_{ins})}{2\pi L k_{ss}} \quad (12)$$

$$R_{convair} = \frac{1}{h_a 2\pi r_{outs} L} \quad (13)$$

where $f_{W/G}$ and f_{air} are the fouling factors. Heat exchangers experience fouling during operation. Surfaces are subject to fouling by fluid impurities, rust formation, or other reactions between the fluid and wall material [32]. Fouling factors are obtained experimentally by determining the values of U for both clean and fouled conditions in a heat exchanger.

$$R_f = \frac{1}{U_f} - \frac{1}{U_{cl}} \quad (14)$$

Here U_f and U_{cl} denote the heat transfer coefficients of the fouled and clean surfaces, respectively. Similar cases can provide fouling factors [33–35]. Fouling also has a significant effect on the pressure drop inside heat exchangers. For a given flow, the pressure drop increases by the fifth power of the ratio of clean to fouled diameter [36].

$$\frac{\Delta P_f}{\Delta P_{cl}} = \left(\frac{d_{cl}}{d_f} \right)^5 \quad (15)$$

where ΔP and Δd represent the pressure drop and changes of the tube diameter due to fouling effects. Equations (14) and (15) indicate that fouling has a significant negative effect on the performance of heat exchangers. A rise of pressure drop increases the required pump power to deliver the same amount of fluid. Hence fouling of the tube bundles increases the internal irreversibilities within the heat engine. The liquid that currently flows through the tube bundles of the MHE is W/G. The heating and cooling systems of the MHE operate in closed loops; therefore, no impurity will be added to the liquid. Furthermore, all tubes are made of stainless steel 304, which does not react with W/G. Compressed dry air is the secondary fluid inside the shell. The compressor that pressurizes the shells separates the water from air that enters the heat

exchangers. In the heat transfer model, the fouling effects on the heat transfer and pressure drops are assumed to be negligible.

Substituting the energy flow of the liquid at the outlet and Eq. (9) into Eq. (8) yields

$$\int_{t_{ini}}^{t_{fin}} \dot{Q}_{in} dt = \int_{t_{ini}}^{t_{fin}} \dot{m}_{W/G,out} \bar{C}_p T(t) dt + UA \int_{t_{ini}}^{t_{fin}} \Delta T(t) dt \quad (16)$$

where $\bar{C}_p$ represents the average specific heat capacity of the liquid during one operational mode and ΔT is the temperature difference between the liquid and gas inside the heat exchanger. This equation gives the heat transfer rate from the fluid inside the tube bundle to the compressed dry air inside the shell.

$$\int_{t_{ini}}^{t_{fin}} \dot{Q}_{ta} dt = m_{air} C_{v,air} \Delta T \quad (17)$$

This equation indicates the increase in the energy of the compressed air during dt . Multiplying Eq. (9) by dt and combining with Eq. (17) yields

$$m_{air} C_{v,air} dT = UA_{tube} (T_{W/G} - T_{air}) dt \quad (18)$$

where T_{air} is the air temperature at any instant time during the charging period. Also, $T_{W/G}$ denotes the liquid temperature through the tube bundle, which is nearly constant.

The W/G thermal capacitance is significantly higher than stored air inside the shells. Also, W/G flows through the tube bundle and compressed air is a stationary fluid. Thus, changes of the liquid and tube surface temperatures are negligible, in comparison to the air temperature changes. Rearranging Eq. (18) and integrating over time from $t = 0$, where $T_{air} = T_{air,ini}$, to a time of t gives the air temperature at any instant in time (t) as

$$T_{air,t} = T_{W/G} + (T_{air,ini} - T_{W/G}) e^{\frac{UA_{tube} - t}{m_{air} C_{v,air}}} \quad (19)$$

This equation indicates that increasing the tube surface area and overall heat transfer coefficient reduces the required time for the air temperature to reach almost the liquid temperature.

An exergy analysis is performed to examine the performance of the MHE under different ambient conditions. In this system, it is assumed that no significant chemical reaction occurs. The physical exergy of each point of the system is determined by

$$Ex_i = (H_i - H_0) - T_0(S_i - S_0) \quad (20)$$

where H_0 , T_0 and S_0 are enthalpy, temperature and entropy of the dry air under ambient conditions, respectively. Moreover, H_i and S_i are the enthalpy and entropy values at each point within each of the heat exchangers.

When the value of $\dot{Q}_{ta}$ is known from Eq. (8), the following equation is used to find the thermal exergy transferred, associated with the heat flow (Ex_Q),

$$Ex_Q = \left(1 - \frac{T_0}{T_{W/G}} \right) \dot{Q}_{ta} = \Gamma \dot{Q}_{ta} \quad (21)$$

where $T_{W/G}$ is the liquid temperature and Γ is the exergetic temperature factor. $\dot{Q}_{ta}$ is the heat transfer rate to/from the compressed air inside the heat exchangers. When heat transfer is maintained to the hot heat exchanger at high temperatures, the exergetic value of the heat flow is higher. On the cold heat

exchanger side, the liquid, which flows through the tube bundles, is at atmospheric temperature. Thus, the exergetic value of the heat flow that is carried by the fluid is zero.

Fig. 5 shows an energy flow diagram of an MHE with four heat exchanger pairs. The overall engine operation is very close to the steady state. For instance, when the first pair starts the charging mode, the second and third pairs exist in operational mode and at the same time, the fourth pair starts the charging mode. Thus, the amount of absorbed/rejected heat from/to the heat source/sink is nearly constant.

4. Energy and exergy efficiencies of Marnoch Heat Engine

Exergy and exergy efficiencies of the heat engine are determined in order to evaluate the performance of the MHE, under various operating conditions. The thermal efficiency (η_{th}) of the heat engine is defined as the net work output to the total energy added by heat transfer,

$$\eta_{th} = \frac{W_e}{Q_{in}} \quad (27)$$

where W_e is the total electric work output during one operational cycle. Q_{in} is the overall heat input that is absorbed by the compressed air during the charging period. The thermal efficiency of the system is not sufficient alone to fully characterize the irreversibilities within the system. For heat engines, Cengel and Boles [37] defined the second law efficiency (η_{II}) as

$$\eta_{II} = \frac{\eta_{th}}{\eta_{id}} \quad (28)$$

Here, η_c is the ideal (Carnot) efficiency of the system as defined below:

$$\eta_c = 1 - \frac{T_C}{T_H} \quad (29)$$

and T_C and T_H indicate the heat source and sink temperatures, respectively. This is the maximum efficiency that any power cycle can have while operating between the same reservoirs.

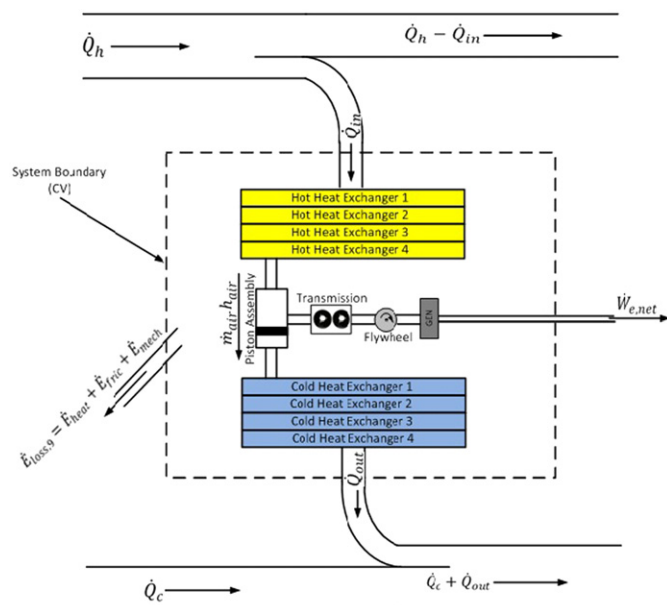


Fig. 5. Energy flow diagram of the MHE.

The MHE generates electricity from low temperature heat recovery. The thermal efficiency does not characterize the effect of energy quality on the thermal efficiency of the system. Therefore, the exergy efficiency (η_{Ex}) of the system is given by

$$\eta_{Ex} = \frac{W_e}{Ex_{in}} \quad (30)$$

The energetic and exergetic values of electricity are equal; therefore, W_e has the same value as Eq. (27). Ex_{in} is the total exergy transferred to the compressed air during one operational cycle. To calculate the exergetic value of transferred heat into the system, the total heat absorbed by the air (Q_{in}) is multiplied by the exergetic temperature factor (T).

5. Results and discussion

In this section, operation of the MHE under various conditions is examined. The effects of the varying heat source and sink temperatures on the energy and exergy efficiencies of the MHE are presented and discussed. Mechanical components and configurations of the MHE units, with higher production capacities, are similar to the current prototype. For instance, the heat recovery capacity can be increased by adding to the number of heat exchanger pairs, transmission systems and generators. Therefore, the results and efficiencies from the current unit are useful for the scaled-up units. Values of the mechanical losses that occur through the piston assembly, bearings, generator and inverter are obtained from the manufacturer specifications. The efficiency of the actuator is about 87% [38]. The transmission system of the MHE is made of six bevel gears, two shafts and seven grooved ball bearings. Experiments show that when the rotational speed of the flywheel is 450 RPM, the generator power output is about 100 W. At this rotational speed, the friction loss of each bearing is about 1.4 W [39]. Thus, the efficiency of the transmission system is estimated to be 90%. The selected generator for the MHE is a PM type generator by Ginlong, with an 85% efficiency [40]. This generator efficiency is close to PM generators with the same rotor alloy as in Refs. [41,42]. The rectifying unit, including a rectifier and diodes, is 94% efficient

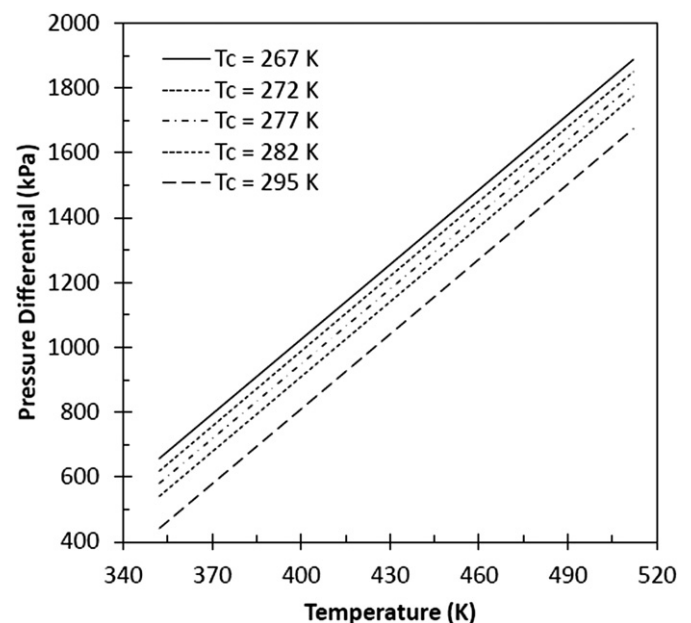


Fig. 6. Pressure differentials between the cold and hot heat exchanger shells at an initial pressure of 1034 kPa.

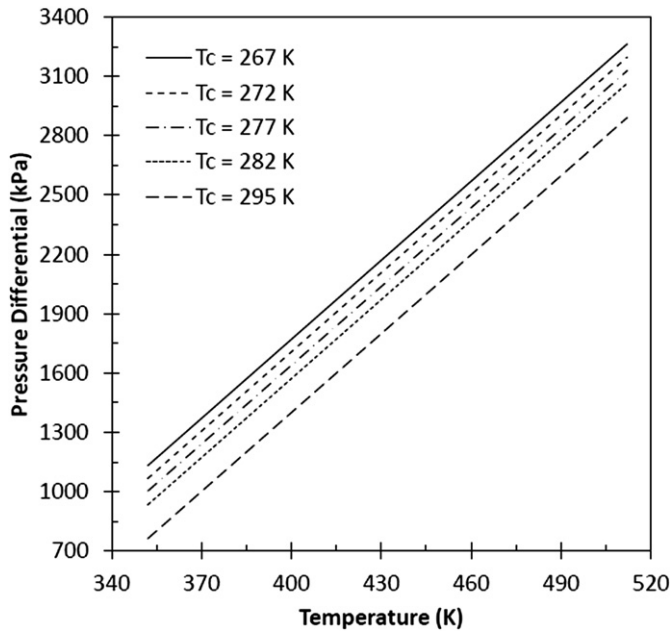


Fig. 7. Pressure differentials between the cold and hot heat exchanger shells at an initial pressure of 1862 kPa.

[43]. These numbers indicate that the overall efficiency of the transmission system and generator, considering mechanical and electrical losses, is about 63%. All the losses, which occur through the transmission system, are mechanical and electrical losses; thus, exergetic and energetic values of losses are equal.

In addition to the mechanical losses that occur through the MHE, energy losses in the form of heat flows are another significant source of energy loss in the MHE. Shells of the MHE heat exchangers are designed to tolerate up to 3447 kPa. The thickness of the carbon steel shells is about 9.5 mm. In the current prototype, the outer layers of the shells are insulated from the outside to prevent significant heat losses from the shells. Since the MHE has a cyclical operation, undesired thermal energy storage occurs

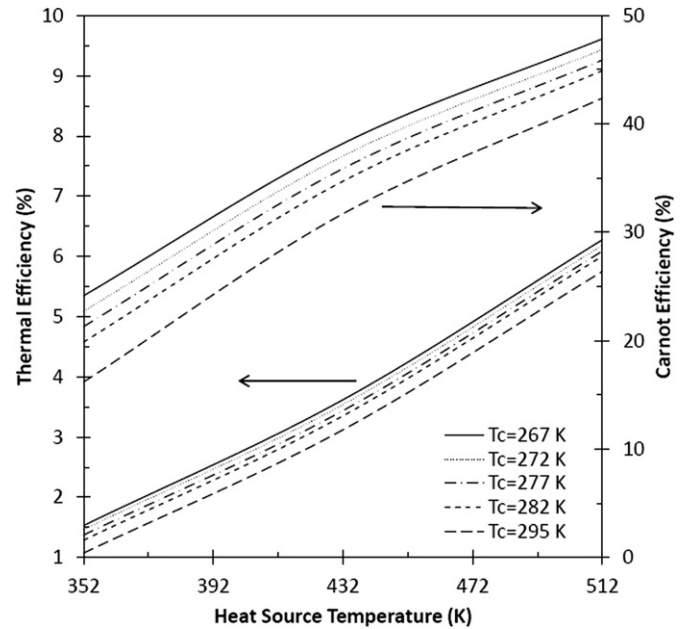


Fig. 9. Variations of Carnot and thermal efficiencies of the MHE at various heat source temperatures ($P_{ini} = 1447$ kPa).

through the shells after each operational cycle. The shells are in direct contact with the working fluid; therefore, during charging and discharging modes, heat transfer occurs between the shell and working fluid.

The stored compressed air mass, inside the heat exchangers, varies with temperature and initial charging pressure. Figs. 6 and 7 show the pressure differentials at the same operational conditions (temperatures) at three different initial pressures. Increasing the initial charging pressure, inside the shells, results in higher mass transfer and consequently higher pressure differentials. Mechanical power output from the piston assembly relates to the piston torque output and speed. Fig. 8 shows the torque output from the piston assembly with two different diameters and configurations (single

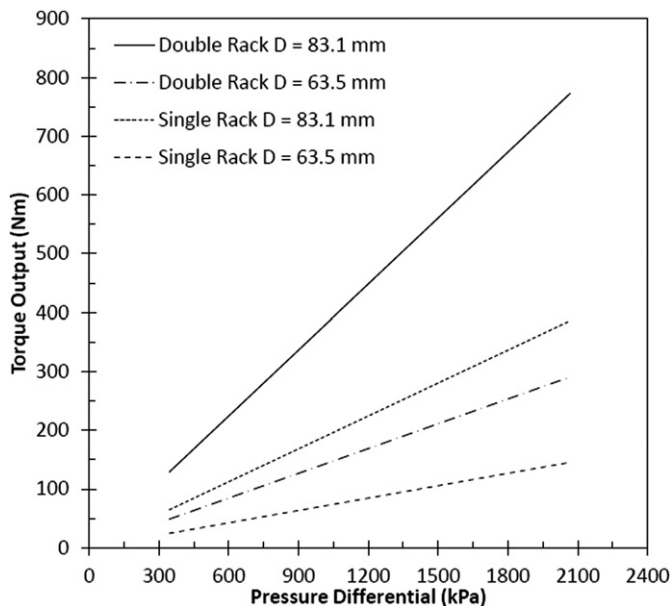


Fig. 8. Torque output from different piston assembly configurations at various pressures.

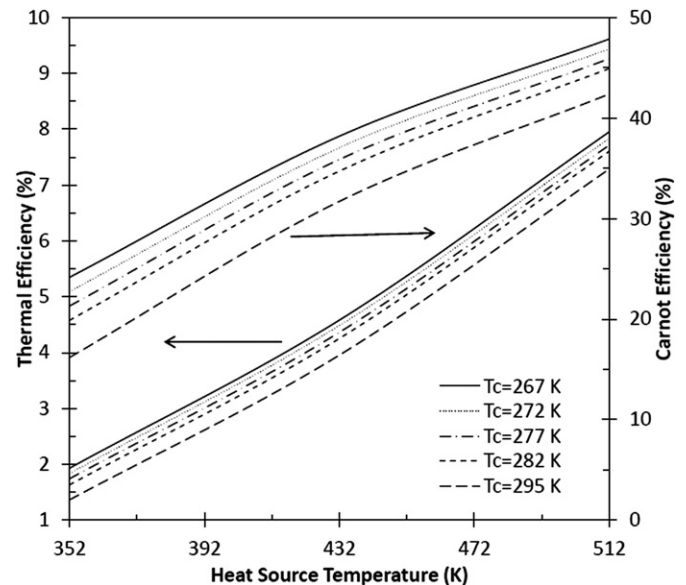


Fig. 10. Variations of Carnot and thermal efficiencies of the MHE at various heat source temperatures ($P_{ini} = 1861$ kPa).

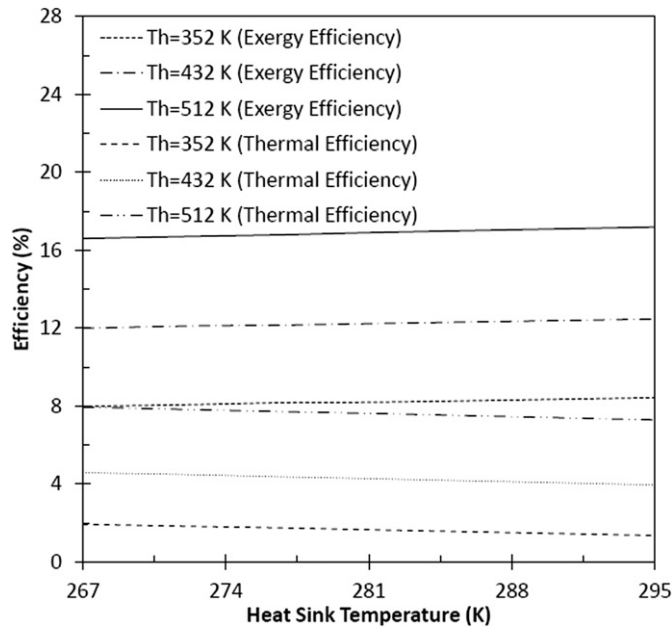


Fig. 11. Variations of thermal and exergy efficiencies of the MHE at various heat sink (T_c) temperatures.

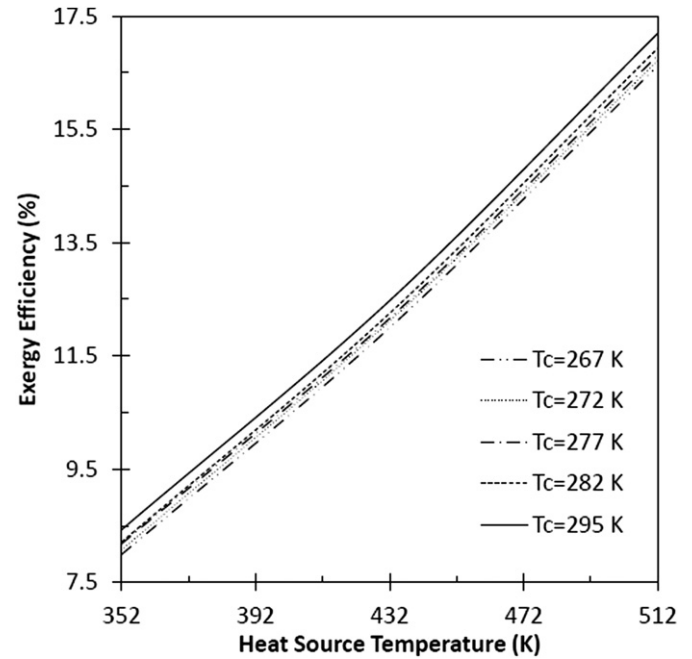


Fig. 13. Variation of exergy efficiency of the MHE at various heat source temperatures ($P_{ini} = 1861$ kPa).

and double rack). Pressure differentials on the opposite sides of the piston assembly depend on the initial charging pressure of the system and the temperature differential between the heat sink and source.

Figs. 9 and 10 show changes of the Carnot and thermal efficiencies under various operating conditions by considering actual cases. Increasing the heat source temperature increases the Carnot and thermal efficiencies of the system. Comparing Figs. 9 and 10 indicate that increasing the initial operational pressure raises the thermal efficiency of the system. However, it has no effect on the

Carnot efficiency of the MHE. Fig. 11 shows changes of thermal and exergy efficiencies of the system at different ambient temperatures (267–295 K). The efficiency variations are less than 1%. At lower temperatures, the overall heat loss from the system increases.

Figs. 12 and 13 show the changes of exergy efficiency with heat source temperature of the system at pressures of 1447 and 1861 kPa, respectively. Increasing the heat source temperature increases the exergy efficiency of the system. Fig. 14 shows the changes of exergy and energy efficiencies of the system with respect to the pressure differentials. The heat sink temperature is assumed to be constant (272 K) and the heat source temperature

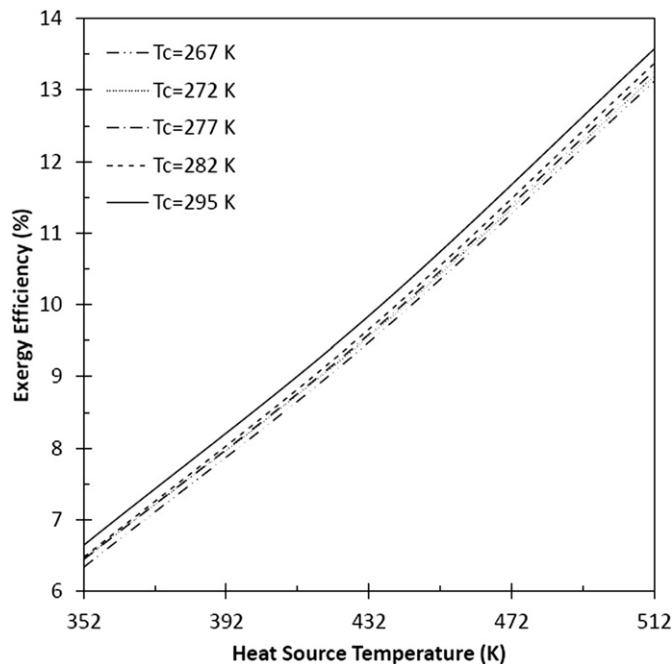


Fig. 12. Variation of exergy efficiency of the MHE at various heat source temperatures ($P_{ini} = 1447$ kPa).

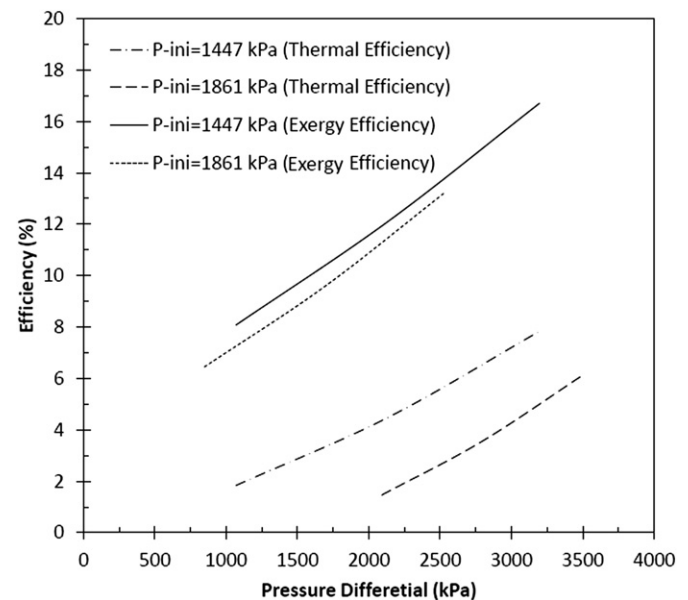


Fig. 14. Variations of thermal and exergy efficiencies of the MHE at various pressure differentials ($T_c = 272$ K, 352 K $< T_h < 512$ K).

varies between 352 and 512 K. This figure indicates that the thermal and exergy efficiencies of the system have higher values when the initial charging pressure of the system is higher. Energetic and exergetic mechanical and electrical losses through the MHE transmission and generator are estimated to be 37%. Comparing the mechanical efficiency of the heat engine to the measured exergy efficiency indicates that up to 80% of the exergy input is lost in the form of heat ($T_C = 272$ K, $T_H = 352$ K). A significant part of heat loss occurs to heat transfer between the air and shells during the cyclic work of the heat exchangers. A possible way to reduce the heat loss, from the air to the shells, is to insulate the shells from inside. The suggested materials for the insulation are silicon rubber or Macor, which is a machinable glass ceramic. These materials have low thermal conductivities; therefore, heat transfer, between the compressed air and shell, during the charging and discharging modes, is not significant.

6. Conclusions

The Marnoch Heat Engine converts low grade waste heat into electricity. The application of the MHE is commercially viable when high amounts of heat flow at a low temperature are available. This study has shown that increasing the operational pressure of the MHE increases the power output from the system. In most industrial processes, waste heat, which is the heat source of the MHE, is constant. Also, the temperature of the heat sink depends on ambient conditions; therefore, it is not normally changed in a predictable manner. The only parameter that can be readily changed is the initial pressure of the shell. Therefore, optimizing the initial operational pressure of the MHE has a direct effect on maximizing power output from the system and increasing the exergy efficiency of the system. When the system is pre-pressurized, it is important to reach almost the maximum design pressure on the hot heat exchanger side during the charging mode. Therefore, a process must be initiated to ensure that the pressure differential on the sides of the piston assembly occurs at a maximum level.

Acknowledgements

The authors acknowledge the financial support and assistance provided by Marnoch Thermal Power Inc. and the Ontario Power Authority.

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